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Considerable progress has, in recent times, been attained in the development of the high-pressure axial blower by well-planned research. The efforts are directed toward improving the efficiencies, which are already high for the axial blower, and in particular the delivery pressure heads. For high pressures multistage arrangements are used. Of fundamental importance is the careful design of all structural parts of the blower that are subject to the effects of the flow. In the present report, several recent results and experiences are reported, which are based on results of German engine research. (See reference 1.)

RAISING OF THE STAGE PRESSURE

In the axial-flow machine in contrast to the centrifugal, the flow medium does not undergo any change in the peripheral velocity u. As a result, there also does not occur any static pressure rise due to the centrifugal force in the blower. The delivery head ΔH(m), or the total pressure rise Δp_{ges} (kg/m^2), expressed nondimensionally by the coefficient

$$\psi = \frac{2g\ \Delta H}{u^2} = \frac{\Delta p_{ges}}{\frac{\rho}{2} u^2}$$

is in the case of the axial blower smaller than in the case of the radial blower; ρ (kg s^2/m^4) in this formula is the density of the flow medium, g (m/s^2) the gravitation acceleration, and u (m/sec) the blade-tip peripheral speed. The striving toward low structural weight and small space requirement and hence favorable installation conditions makes an increase in the pressure per stage necessary. The possibility of raising the blower pressure by increasing the rotational speed is limited because,

*"Neuere Erfahrungen an Überdruck-Axialgebläsen." V.D.I. Zeit. vol. 88, no. 37/38, Sept. 16, 1944, pp. 516-520.

in addition to the undesirable increase in the blower noise and operational difficulties, the effect of the air compressibility comes into evidence as the relative velocities approach the velocity of sound. If the notation used for airfoil theory is carried over to the similarly shaped and similarly acting blower blades, the effect of the air compressibility upon increasing the Mach number, that is, the ratio of the flow to the sound velocity, is expressed in a decrease in the lift coefficient c_a with simultaneous increase in the drag coefficient c_w and therefore the drag-lift ratio $\epsilon = c_w/c_a$, which affects the efficiency, is impaired.

As numerous tests (reference 2) undertaken by the Reich Communication Ministry have shown, however, the lift coefficient of each blade and therefore the pressure coefficient can be effectively increased by strongly cambering the blades. As an example, figure 1 shows the characteristic curves of a single-stage axial blower with strongly cambered blades. The pressure coefficient ψ and the blower efficiency η_G for various settings of the blades are plotted against the dimensionless coefficient

$$\varphi = \frac{c_m}{u} = \frac{V}{Fu}$$

which characterizes the flow delivered per unit time where c_m (m/s) is the meridional velocity, that is, the component of the absolute flow velocity c in the direction of the axis, V (m^3/s) is the volume delivered per second, and F is the blower cross section. Through a corresponding initial rotation produced in a guide-vane apparatus connected ahead of the impeller, for example, under the assumption of $\psi = 1$ and air delivered under normal conditions (0° C, 760 mm Hg), increases in pressure in a single stage of 5000 to 6000 millimeters of water may be produced. If it is desired in this manner to attain high-stage pressures and best efficiencies, the Reynolds number, corresponding to the mean relative inflow velocity and the length of the blade at the outer diameter, must in no case be smaller than 100,000.

STRUCTURAL INFLUENCES

In addition to the aerodynamical requirements, the structural design of the individual blower parts affects the attainable maximum pressure and the efficiency.

Effect of the Inlet Duct

In order to avoid losses during the inflow of the flow medium into the blower, an inlet duct is necessary. The velocity distribution ahead of the blower depends on the aerodynamic quality of the inlet duct. Tests with and without variously shaped inlet ducts led to the results shown in figure 2. As is seen, the velocity distribution ahead of the blower and therefore the impinging on the blower is considerably less uniform without an inlet duct than for the case of a correctly shaped duct. According to investigations outside this country, the pressure coefficient in the working range of the blower may, by means of inlet ducts, be increased by 10 to 20 percent and the efficiency by 10 to 15 percent (fig. 3).

Effect of the Hub Shroud at the Upstream Side

With the development of the axial blowers toward high pressure coefficients, the shape of the hub shroud on the upstream side has acquired added significance. An increase in the pressure coefficient according to these results presupposes an increase in the hub ratio $\nu = d/D$, where d is the hub diameter and D the outside diameter of the impeller. The shaping of the hub shroud on the upstream side affects the blower-characteristic curves, that is, the outputs and blower efficiencies. As shown by the results in figure 4, a hub shroud is advantageous for every operating condition. For the four shapes of hub shrouds investigated, the differences are relatively small. The most favorable was found to be the semispherical hub shroud because for it the sum of the separation and friction losses are presumably smallest. The curves of constant throttle number σ characterize the ratio of the dynamic pressure of the flow to the total pressure increase

$$\sigma = \frac{\frac{\rho}{2} {c_m}^2}{\Delta p_{ges}} = \frac{\varphi^2}{\psi}$$

Effect of the Hub Shape at the Downstream Side

The pressure conversion behind the blower is greater the greater the loading and therefore the hub ratio.

The total pressure to be converted by the blower is

$$\Delta p_{ges} = \sum \frac{\rho}{2} v^2 \zeta + \frac{\rho}{2} c_{m_1}^2 - \Delta p_{stat} \ (\mathrm{kg/m^2})$$

where v (m/s) is the velocity at the point considered and c_{m_1} (m/s) the meridional velocity in the blower. The first term in this expression represents the total pressure required to overcome the resistances in the pipe, characterized by the resistance coefficient ζ, $\frac{\rho}{2} c_{m_1}^2$ is the kinetic energy communicated to the air, and Δp_{stat} the static energy recoverable in a diffuser.

By introduction of an ideal over-all efficiency of the diffuser

$$\eta_{id} = \frac{\frac{\rho}{2}\left(c_{m_1}^2 - c_{m_2}^2\right)}{\frac{\rho}{2} c_{m_1}^2}$$

where c_{m_2} (m/s) is the outflow velocity from the diffuser into the pipe, there is obtained for the required total-pressure increase

$$\Delta p_{ges_{id}} = \sum \frac{\rho}{2} v^2 \zeta + \frac{\rho}{2} c_{m_1}^2 \left(1 - \eta_{id}\right)$$

By taking into account the efficiency of conversion in the diffuser through the efficiency η_D, there is obtained

$$\Delta p_{ges} = \sum \frac{\rho}{2} v^2 \zeta + \frac{\rho}{2} c_{m_1}^2 \left(1 - \eta_{id}\ \eta_D\right)$$

Because the ideal over-all efficiency of the diffuser is affected only by the structural magnitudes, the most suitable diffuser shape can be obtained by a very simple consideration. Three structural forms enter into consideration (fig. 5):

Form

a The diffuser with increasing outer diameter and constant hub diameter

b The diffuser with increasing outer and hub diameter

c The diffuser with constant outer and decreasing hub diameter

By using for the structural forms a and c an angle of 8° that was found useful in practice, the practical external angle $\gamma_a = 16°$, and the hub angle $\gamma_i = 12°$, there is obtained, for the length to diameter ratios $l/D = 0.5$, 1, and 3, the dependence of the ideal over-all efficiency of the diffuser on the hub ratio of the axial blower (fig. 5). For the axial blowers with hub ratios $\nu > 0.5$, which is of particular importance for high loadability, diffuser shapes of type b therefore appear most promising.

Tests are in process for obtaining the efficiency η_D of the internal conversion. As has already been shown by Wendt (reference 3), for hub diffusers (structural shape c) the internal conversion efficiencies η_D lie at about 88 to 89 percent. Similar results may also be expected according to the tests so far conducted also for the diffuser shapes a and b. The improvement attainable through the diffuser is seen with particular clearness in figure 6, which shows the blower outputs and the blower efficiencies with and without a short-ring diffuser for the three different blade settings.

EFFECT OF UTILIZING ONLY PART OF THE IMPELLER BLADES ON THE CHARACTERISTICS AND EFFICIENCY

Certain structural requirements, for example bearing base and piping particularly in the case of cooling blowers for vehicles, often permit only part of the axial impeller to be utilized as a result of which the characteristic curves, that is, the air outputs, change. Through the friction of the impeller and the flow at the blades not utilized by the flow medium, losses arise that lead to a reduction in the attainable blower efficiency. Tests with sector-shaped covers of the impeller gave the change in the performance and efficiency curves shown in figure 7 for an axial blower with the utilized sector angle δ between 10° and 360°. A further series of tests with a segment-shaped covering led to similar results as with the sector covering.

These tests show that both the blower outputs expressed by the pressure coefficient ψ and the delivery coefficient φ, as well as the blower efficiency η_G for small impinging area, decrease with relative rapidity, a fact that must be taken into account in designing a partly utilized impeller.

It is possible, however, under certain conditions through partial impeller impingement to obtain better blower outputs, namely, when the operational point for the fully utilized impeller would already lie in the unstable part of the characteristic curves. According to test results thus far obtained, a throttle number $\sigma = \varphi^2/\psi = 0.1$ should for this reason not be used.

Assume, for example, that the problem was for a prescribed blower speed and given blower outer and inner diameter to deliver a prescribed quantity of air with a definite blower pressure. The pressure coefficient was $\psi = 0.6$, the delivery coefficient $\varphi = 0.232$, and therefore the throttle coefficient $\sigma = 0.09$. Good efficiencies would not be expected here if the operating point did not already lie in the unstable "surging" stage. Through an uncovering of 360° to 310°, there was obtained a throttle number $\sigma = 0.122$ and according to figure 7 an efficiency of 75 percent with the assurance that the prescribed air quantities are attained. Figure 8 shows the designed partly utilized impeller.

EFFECT OF THE BLADE SHAPE AND OF THE SURFACE ROUGHNESS

In the design of an axial blower with large outputs, the effect of the blade shape, for reasons of economy and saving of material, is to be taken into account. In order to produce aerodynamically favorable blade sections, high-quality blade cutting machines are required or spray cast parts of metal alloys may be used. For many application purposes it is, however, more economical and technically feasible instead of aerodynamically favorable profiles to use a welding construction and provide blades that can be produced relatively simply from pressed-sheet parts. Figure 9 shows test results on a 12-blade axial blower with light-metal profile blades. With a 12-blade impeller but with sheet-metal blades, which are considerably simpler and cheaper to produce, the line of curvature of the blade corresponding to the skeleton line of the cambered profile blade, practically the same outputs and efficiencies can be attained (fig. 9) as with the first-mentioned blower.

For high-value axial blowers, the degree of surface smoothness of the blades is likewise of importance because the friction losses depend on it. It is to be noted here that with increasing Reynolds number higher requirements must be set for the surface smoothness. Particularly in the case of wide blades, it is necessary to observe good smoothness conditions in order to obtain a favorable efficiency. For a qualitative consideration of this problem, the test data of figure 9 may be used. At the operational speed of this blower, the Reynolds number corresponding to the mean relative approach velocity and the blade section length at the outer diameter was 325,000.

The curve for the blower denoted as "rough" was obtained after a first scrubbing of the newly turned impeller, the height of the roughnesses being about 1/10 millimeter. After polishing there was obtained the curve denoted as "smooth." The difference, as is seen, is only slight. In addition the "smooth" blower was produced as a model from commercial cast iron where for technical reasons in connection with casting the desired sharp trailing edge could not be realized and had to be replaced by a rounded edge of about 2.5-millimeter radius. There was now obtained, as a result of the increased roughness and the rounded trailing edges of the blades, the curve likewise shown in figure 9. The deviation is here considerably greater.

Also due to the increased masses and the required greater acceleration forces in starting, axial blowers with cast-iron blades should have limited application.

VOLUMETRIC EFFICIENCY

In connection with the required accuracy in designing the axial blower, the question arises frequently of the effect of the radial clearance on the blower outputs. As tests have shown, there exists, according to results thus far obtained, the relation shown in figure 10 between the efficiency and the relative clearance. This relation may be expressed by the following empirical expression:

$$\eta_{vol} = e^{\frac{-2\ s/D}{\sigma}(3.1 - 8.1 \cdot 2\ s/D)}$$

where e is the basis of Naperian logarithms, s the clearance width, D the impeller diameter.

The use of a cover band, as is usual with steam turbines, leads to more favorable strength relations for the blades but, in spite of a certain labyrinth effect, gives no improvement in the volumetric efficiency (fig. 11).

SUMMARY

Axial blowers are suitable particularly for delivering large air quantities and may readily be built into delivery piping inasmuch as their housings themselves constitute piping elements. Together with high efficiencies, there are also attained high pressures, namely 5000 to 6000 millimeters of water per stage. This performance presupposes, however, a careful air-flow design of the individual parts of the blower. Thus by using inlet ducts, the velocity distributions can be improved and the pressures and efficiencies therefore raised.

The hub shrouds, which on the downstream side form diffusers with the surrounding housing, led to closer investigations on the effect on the blower characteristics and the conversion of the flow energy into pressure. On the upstream side, the semispherical hub form appeared the most favorable. On the downstream side, for hub ratios $\nu > 0.5$, the diffuser with increasing outer and hub diameter appeared to be promising (fig. 5, form b).

Partial utilization of the impeller by covering a segment of the latter may lead to better blower performance if the operational point of the fully utilized impeller is in the unstable range.

An empirically derived equation gives the effect of the radial clearance on the blower output, the relation giving the volumetric efficiency as a function of the relative clearance.

Investigations on a 12-blade blower wheel with well-constructed light-metal streamlined blades and with pressed-sheet blades, respectively, gave practically equal efficiencies and outputs. Considerably cheaper designs can therefore be obtained by welded constructions with pressed-sheet metal blades.

Translation by S. Reiss,
National Advisory Committee
for Aeronautics.

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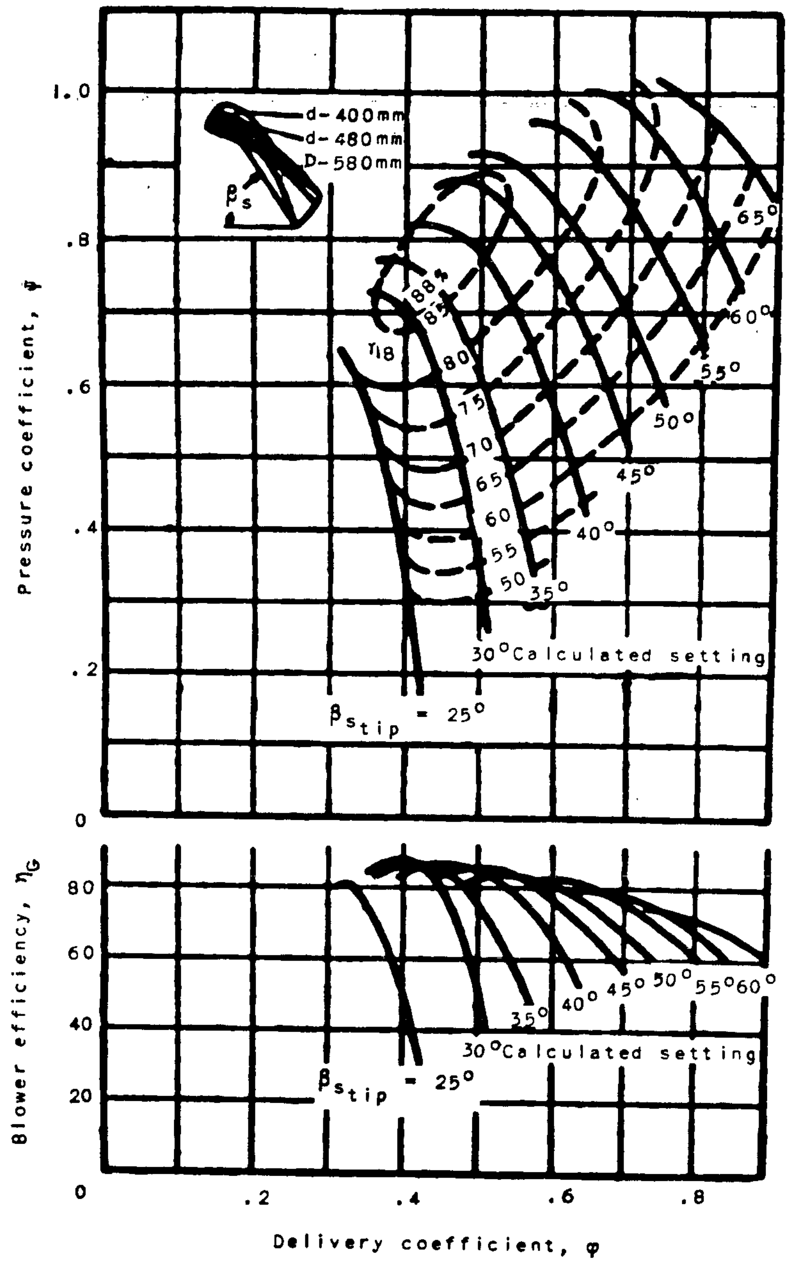

Figure 1. - Characteristics of high-pressure axial blower. β_s, blade angle corresponding to chord of profile skeleton line.

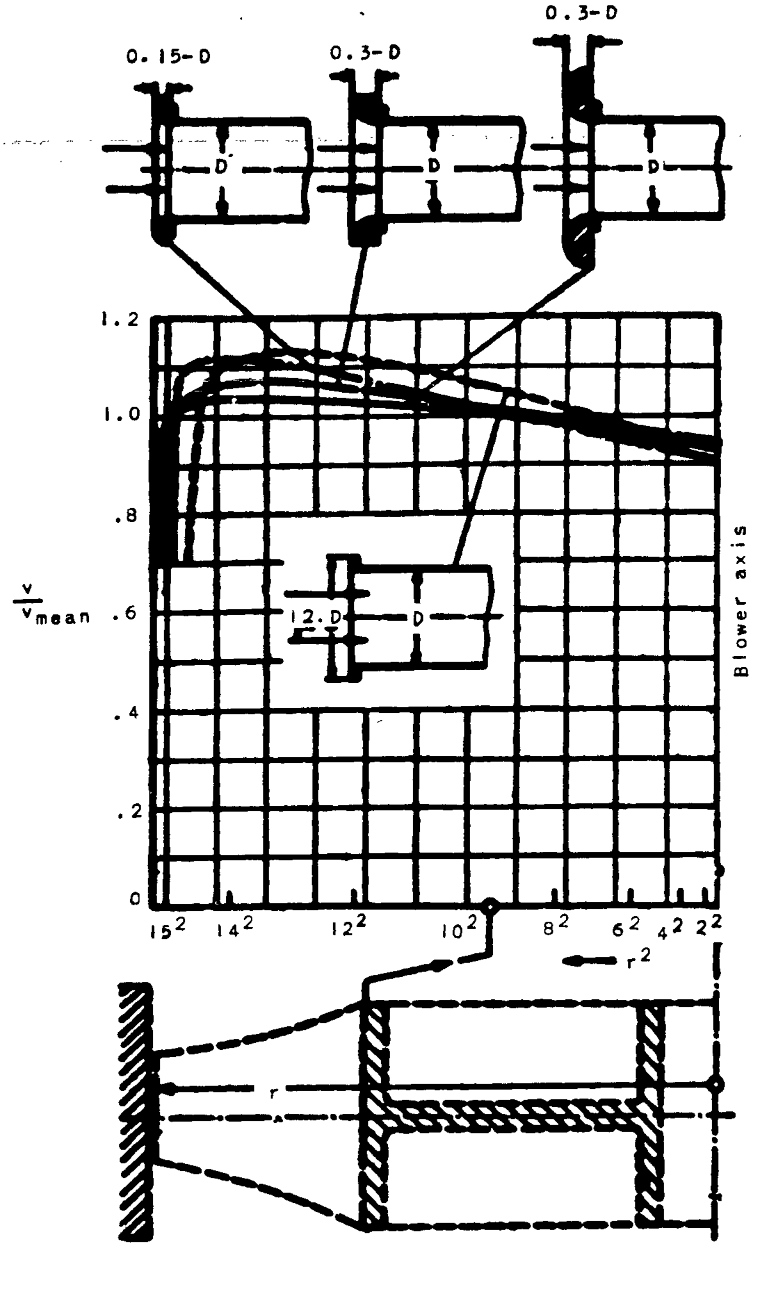

Figure 2. - Effect of inlet-duct shape on distribution of velocity v of axial blower plotted against square of impeller radius r = 0.5 D. v, local (measured) velocity; v_{mean}, mean velocity. The velocity distribution was measured directly behind inlet duct.

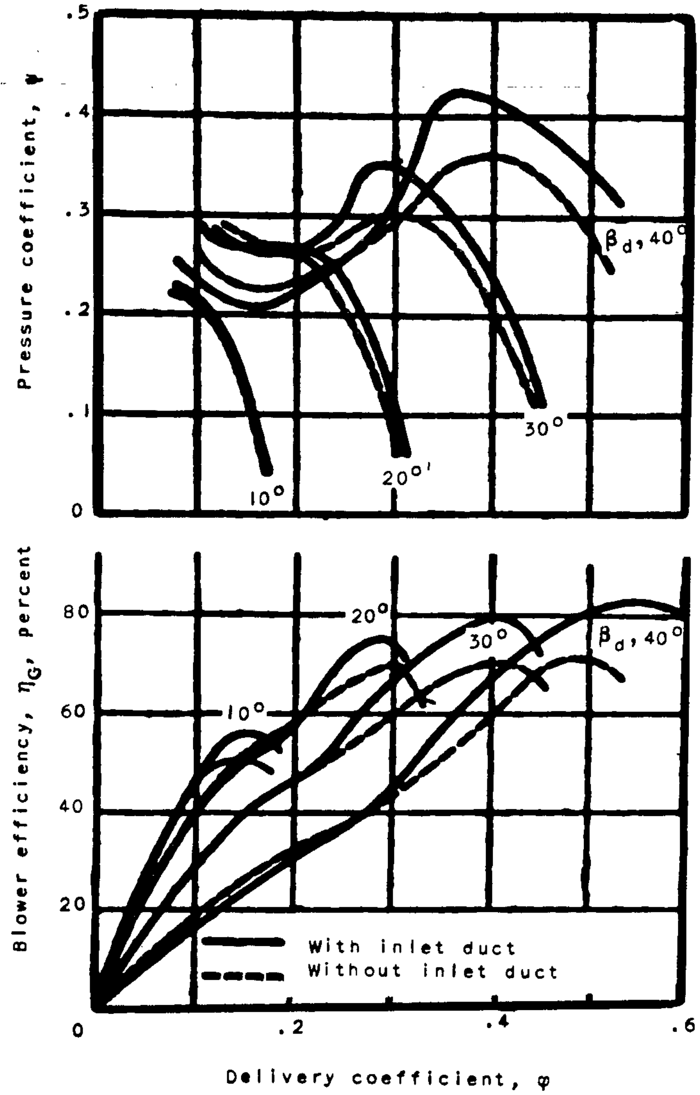

Figure 3. - Change of blower curves when inlet duct is missing. β_d, blade angle on pressure side.

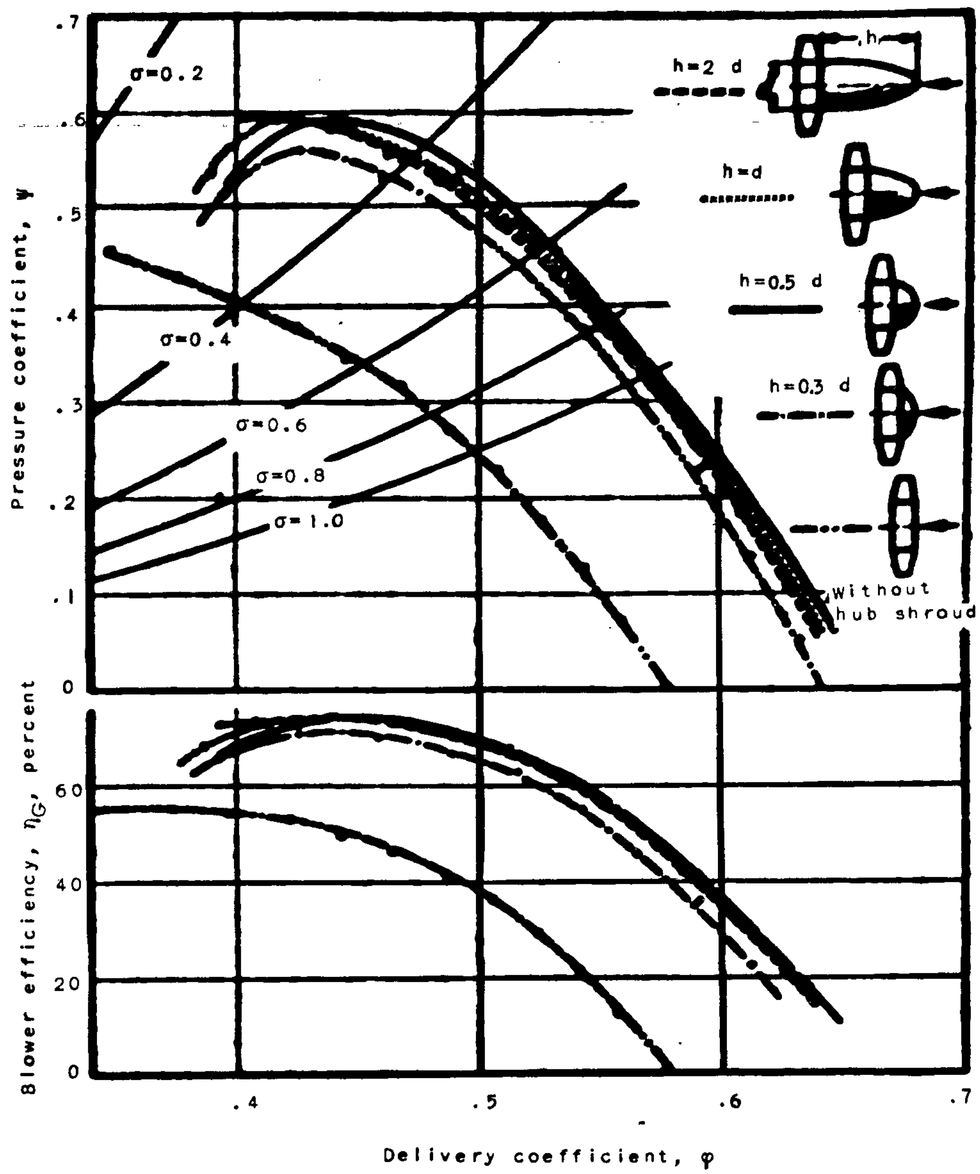

Figure 4. - Effect of hub shroud on upstream side on blower output and blower efficiency.

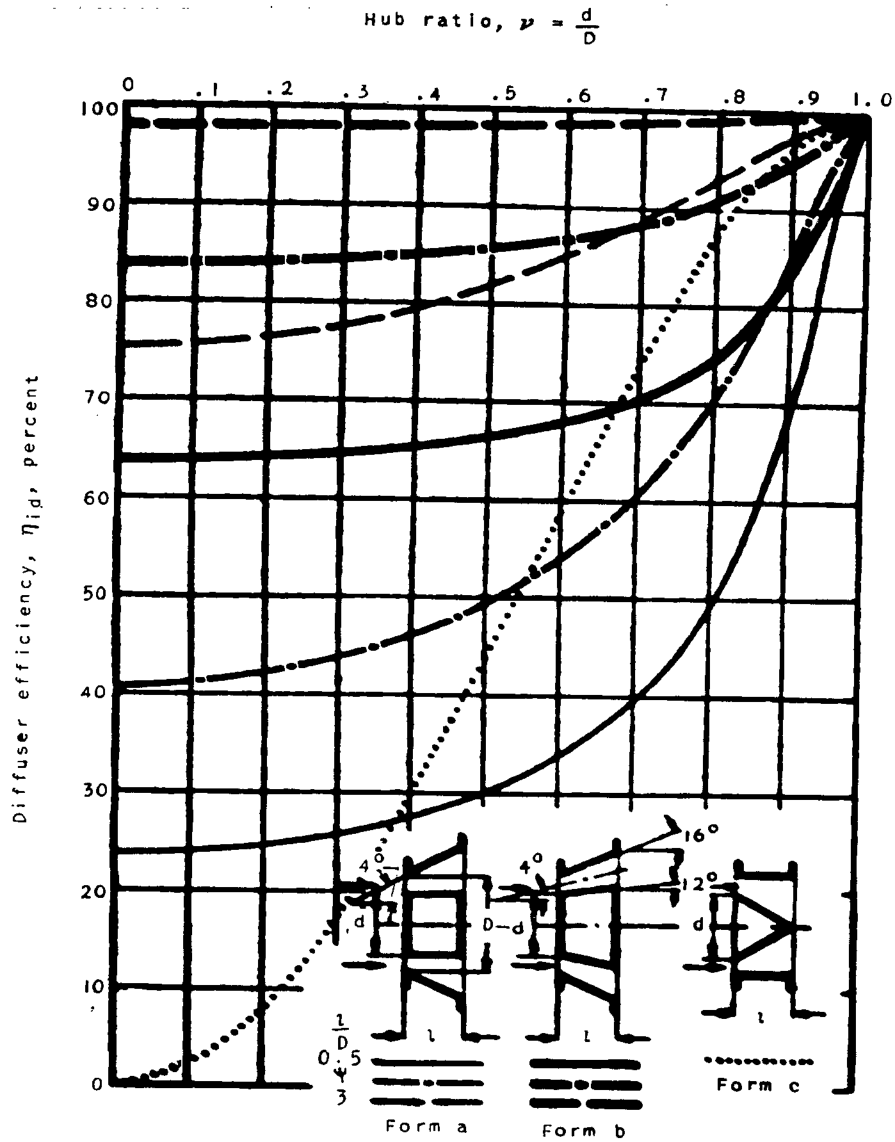

Figure 5. - Dependence of ideal diffuser efficiency on hub ratio for various shapes of downstream side of blower.

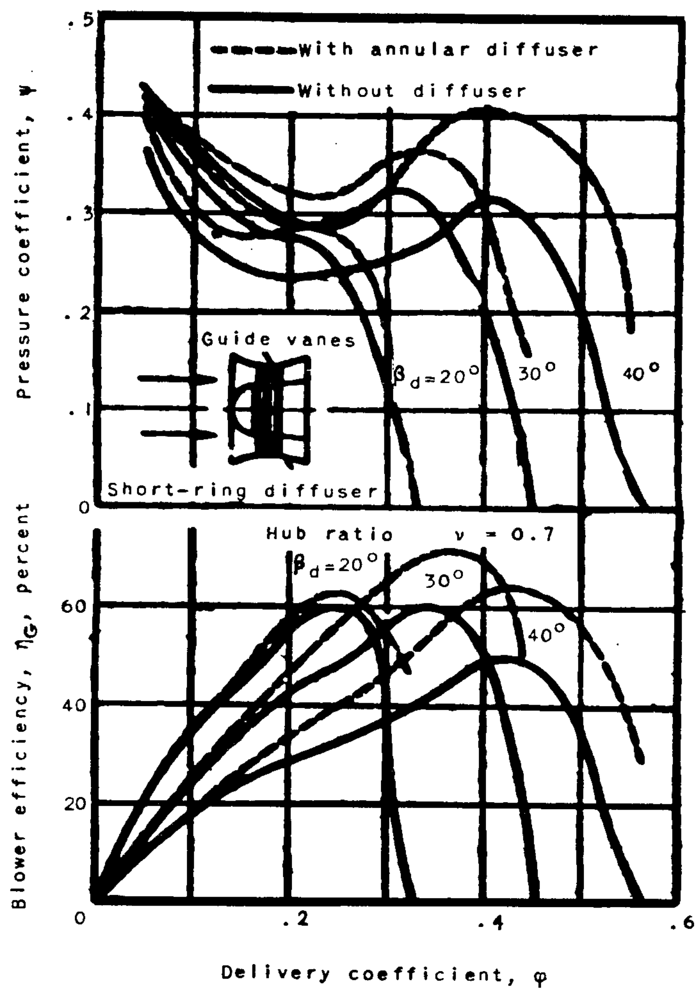

Figure 6. - Blower output and blower efficiency with and without short-ring diffuser. β_d, pressure side angle.

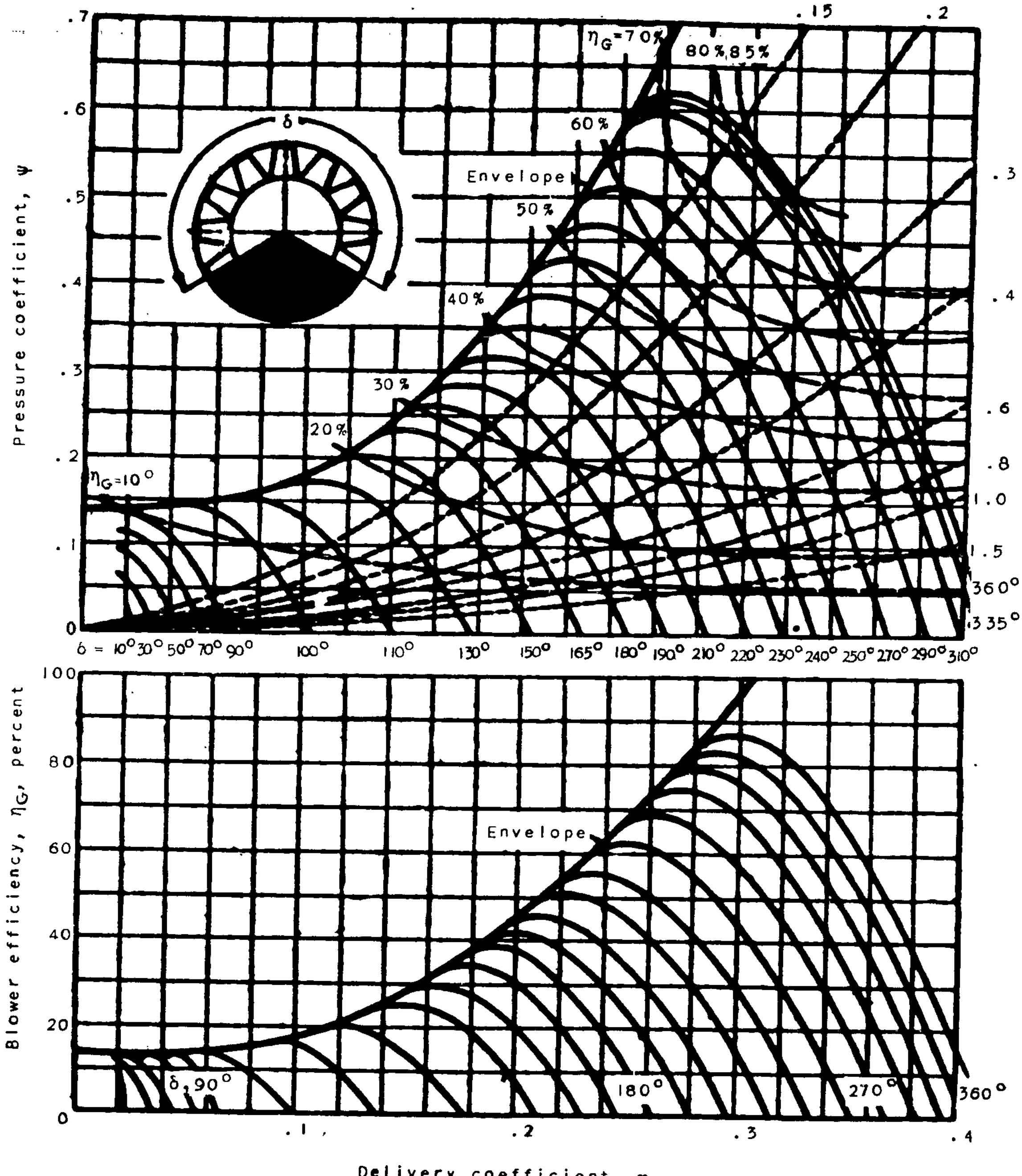

Figure 7. - Change of blower output and blower efficiency on covering a sector of impeller.

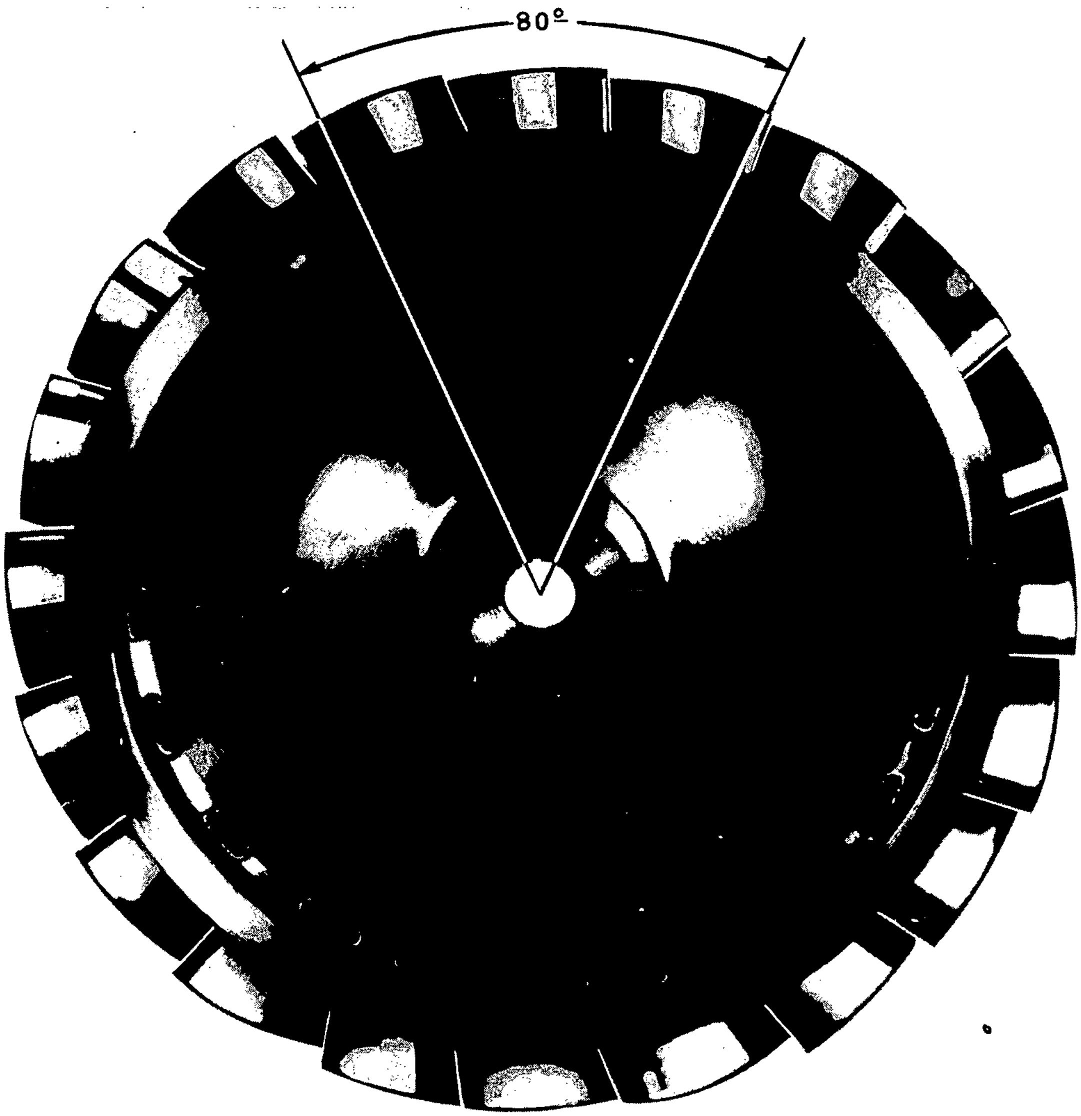

Figure 8. - Axial blower for partial utilization of impeller.

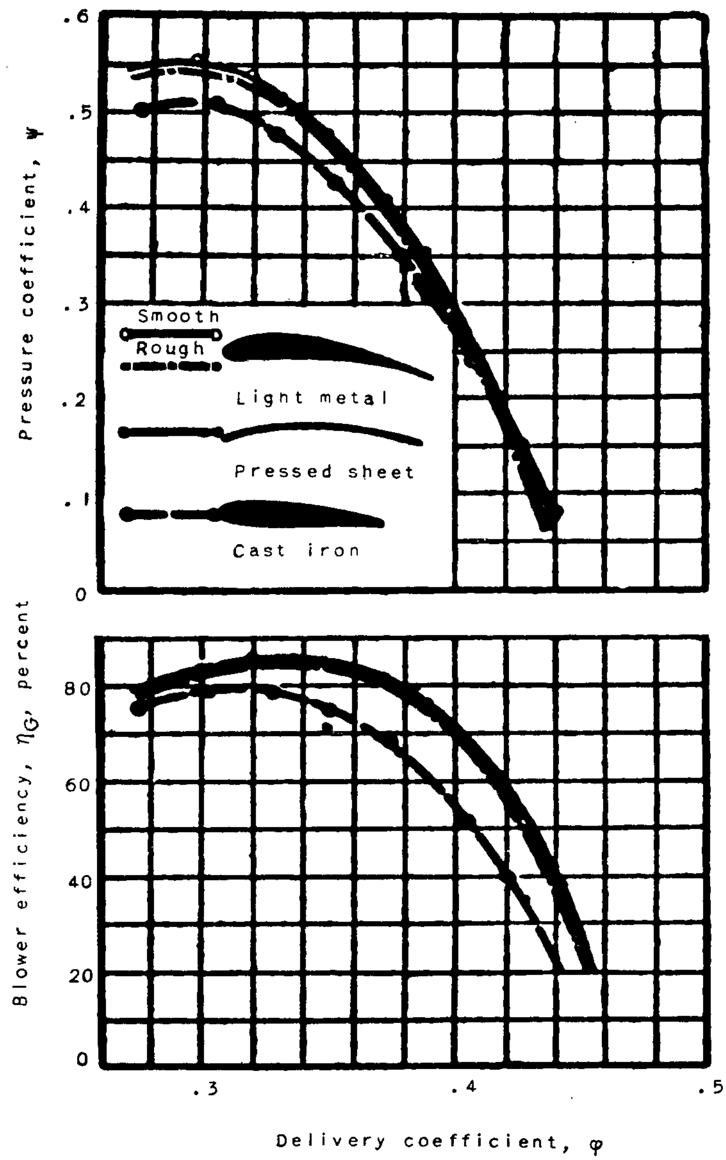

Figure 9. - Effect of blade shape and surface quality on output and efficiency of axial blowers.

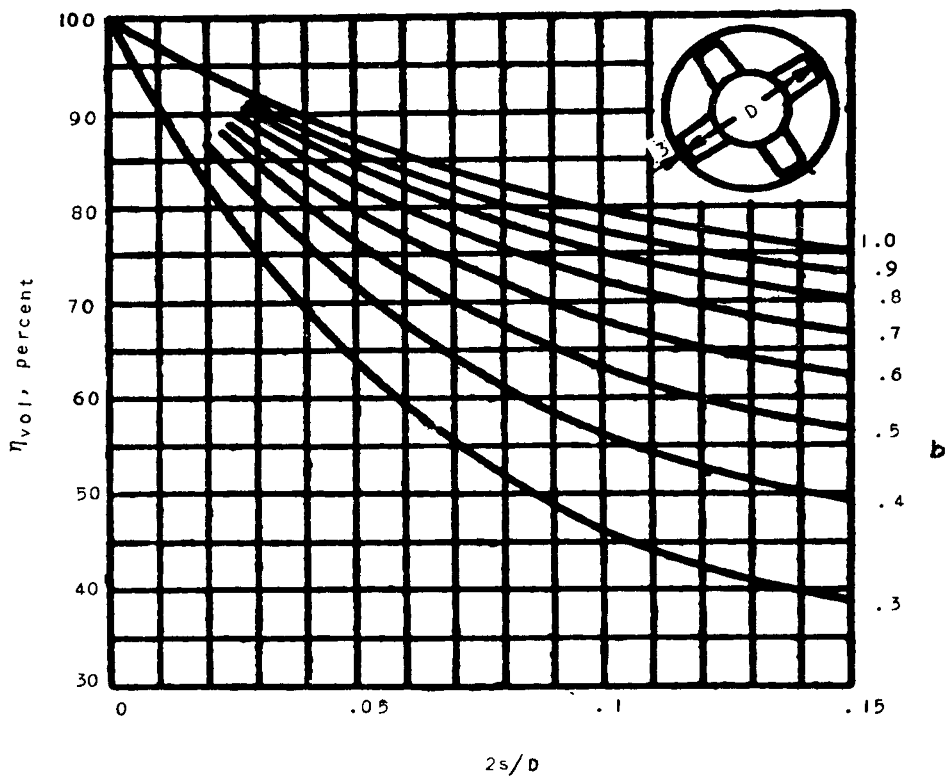

Figure 10. - Effect of impeller clearance on volumetric efficiency of axial blower for various throttle factors. $\sigma = \varphi\ 2/\psi$.

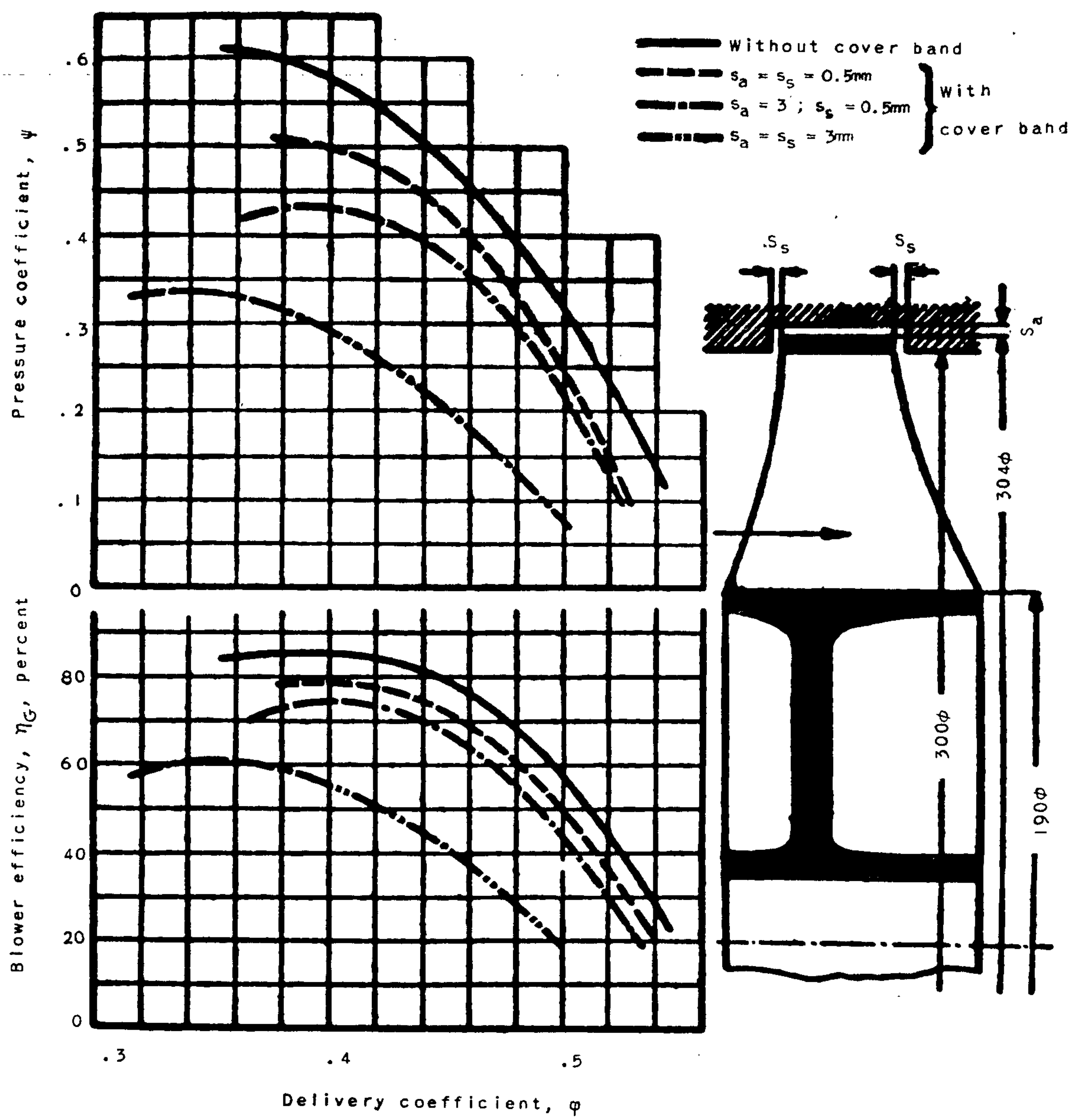

Figure 11. - Effect of outer clearance s_a and of lateral clearance s_s on output and efficiency of axial blower with and without cover band.